Latest Results in the CVT Development

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Introduction

The main requirements of the drive trains for the future are defined: Further fuel saving with simultaneous increase in driving dynamics, while maintaining at least the same level of comfort. These aims must be achieved with increasing engine torques but also without compromise to costing strategy.

The multitronic® transmission from Audi, featuring major components produced by LuK, presents a very good solution for the above mentioned requirements. Also, during the development of this project, there was a requirement for increases in torque capacity. The original transmission arrangement envisaged 250 Nm, meanwhile 310 Nm units are in production, with further torque increases about to be introduced.

This article will address three main points of the CVT development:

- Increasing torque capacity
- Improving efficiency
- Increasing acoustic comfort

Torque Capacity

For the variator design of new engine/transmission combinations with higher torque capacity a suitable load cycle is necessary. On mid and high-range engine concepts, it is not necessary to scale the load cycle proportionally higher with the engine power, as the engine is often driven under partial load. The evaluation of various customer groups reveals the relationship between the average power at the wheel (load cycle) and the nominal engine power, as shown in figure 1.

Depending on the customer's load cycle, there is a bandwidth of average wheel power over engine power. Averaging of the bandwidth, in the range of engine power to be investigated, indicated that with a 50% increase in nominal engine power, the average wheel power only increases by around 30%. The loads analysed from the customer's measurements must now be converted into variator and chain loads. This takes place based on various calculation processes.

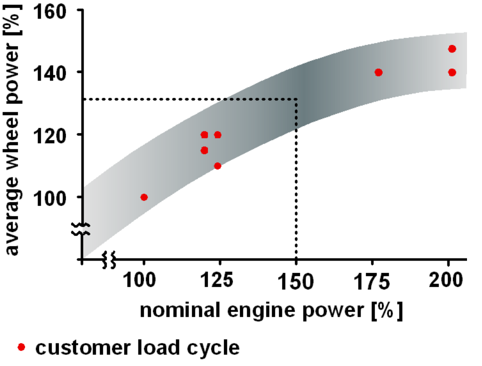


Fig. 1: Influence of Engine Power on Given Average Wheel Power in Vehicle

To increase the torque capacity, the following two main points are considered:

- Strength-optimisation of chain links
- Influence of 'link plate pattern'

Strength-Optimisation of Chain Links

The optimisation takes place by means of classic FE calculations. The aim is to increase torque capacity while simultaneously reducing the mass and the overall size of the chain.

Firstly, the chain link plate forces are determined. This takes place using the simulation tool CHAIN which can consider up to 900 degrees of freedom. CHAIN very accurately calculates the dynamic link and rocker pin element forces to be expected. On the basis of this calculated link plate force, the stress distribution for that particular contour is determined.

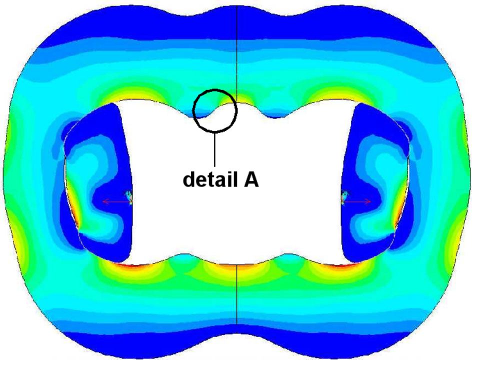


Fig. 2: *Link Stress Distribution*

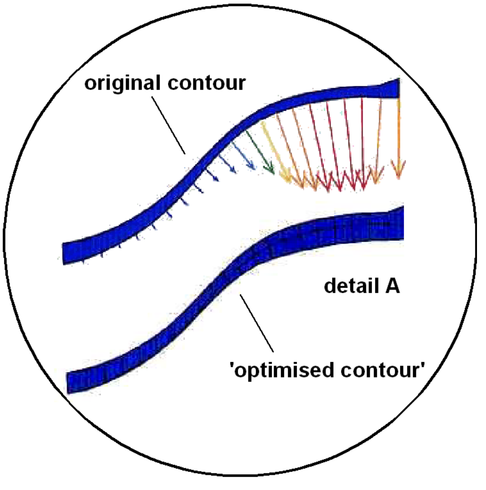


Fig. 3: *Contour Formation According to Biological Growth*

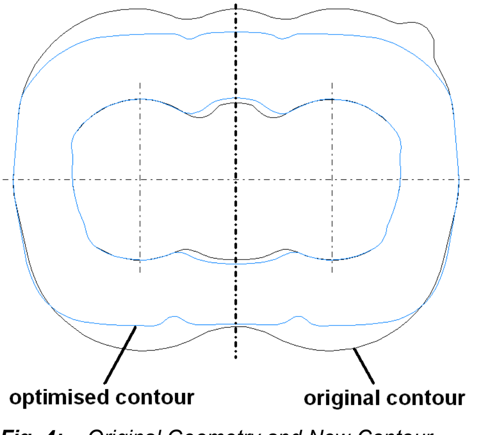


Fig. 4: *Original Geometry and New Contour*

In the following optimisation, the calculation program uses the principle laws of growth, as they exist in nature (bionic method).

The basic principle of this method is the removal of material in areas of low stress and the addition of material in areas of high stress.

Figure 4 compares the original and the optimised link geometry. The new geometry, referred to in the following as 'light-variant', has a 17% reduced mass and a 15% lower radial dimension. Additionally, the maximum stress reduces by around 5%.

The significantly lower mass and the associated lower centrifugal forces have a very positive effect on the link plate stresses (-5%) at high speeds.

On today's chain variator arrangements, the chain is no longer the determining factor for the minimum possible running radius, the shaft strength is. The radial space gain from the link optimisation is therefore used to increase the shaft diameters. Thus, the necessary stress reduction of the shaft is achieved without reducing the ratio spread.

The results of the computer-optimised link plates are meanwhile verified during trials. For this, the link plates undergo standard pulse tests. The running times of the optimised chain contour are around 1.3 times greater than that of the original contour.

Link Plate Pattern of Chain

Trials on test benches and in vehicles unanimously show that with the current chain designs there are still available reserves regarding loading of the rocker pin elements. This potential can be exploited by varying the link plate pattern, i.e. the arrangement of link plates across the width of the chain. This should further reduce the loading on the links, whereby the loading on the rocker pin elements is increased. The main aim is to find a link plate pattern whereby both chain links and rocker pin elements are proportionally loaded and consequently also exploited to the maximum.

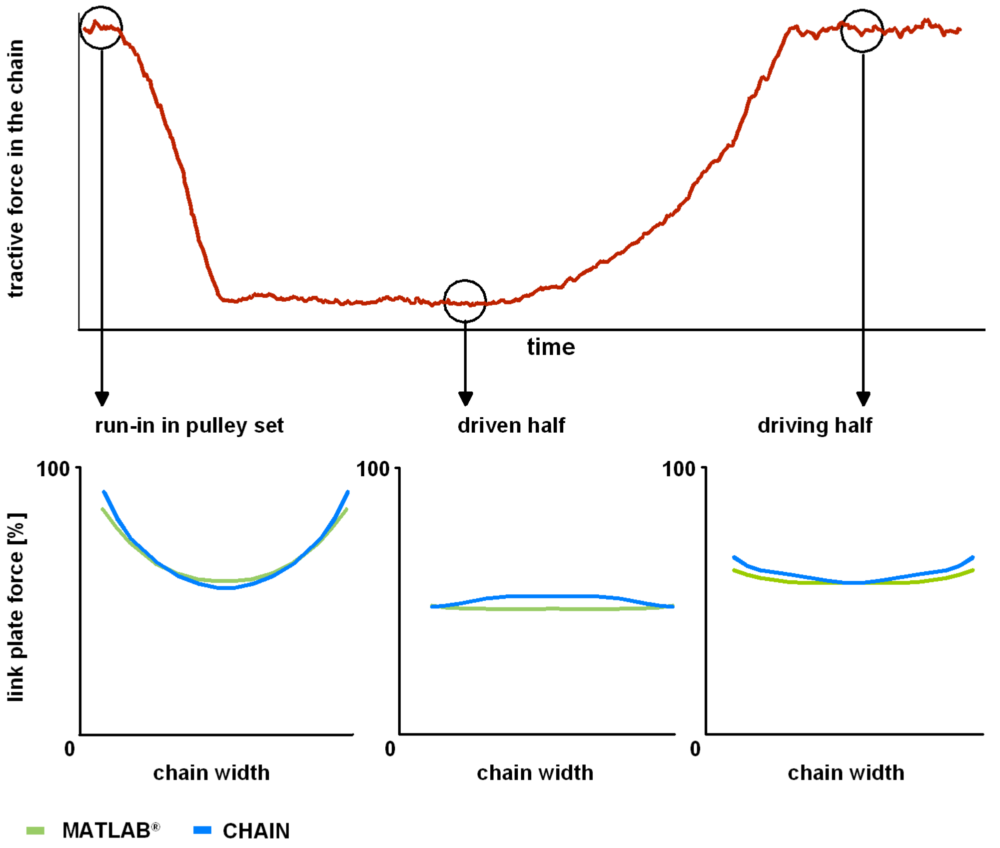


Bild 5: Comparison, MATLAB® and CHAIN Calculations at Three Different Load Points

The number of possible link plate patterns can be significantly limited by setting necessary marginal conditions. The previously mentioned CHAIN calculation program is not considered viable for evaluation of the various conceivable variations because of its processing time. Therefore, a simplified calculation program based on **MATLAB®** has been developed for the pre-selection process.

Figure 5 shows that the qualitative consistency of results is good, and therefore this is a permissible simplification.

Only a few link plate patterns, on which the component loading in total was minimal, are built as prototypes and tested.

The running times of the chains with optimised link plate pattern are around 2 times greater than those of the original variants (figure 6).

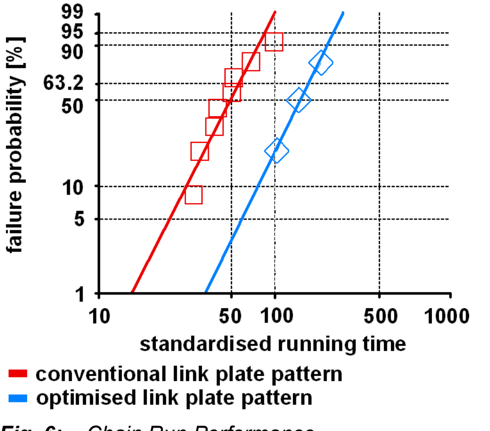


Fig. 6: Chain Run Performance

Torque Capacity, Conclusion

Along with the mechanical optimisation of the chain, there are further reserves in the application system for increasing the torque capacity. As shown in [2], there is a potential of around 5% through optimisation of the clamping system.

Taking all the specified measures into consideration, the torque capacity can be increased by around 30% compared with current production standards. Figure 7 illustrates this relationship.

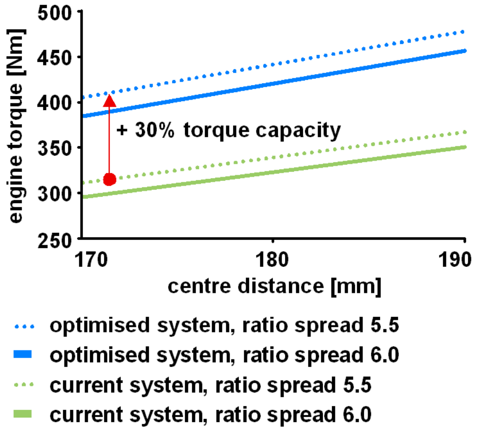


Fig. 7: *Torque Capacity of Chain Variator*

Improving Efficiency

The CVT was originally promoted due to its ability to run the engine at either its most consumption-optimised or performance-optimised operating modes. Meanwhile, further advantages such as high levels of driving comfort and outstanding driving dynamics have emerged as arguments for the CVT. Following the successful introduction of CVT transmissions for higher torque ranges, the focus is now on further optimisation of the efficiency.

The CHAIN program is the basis for the efficiency calculations.

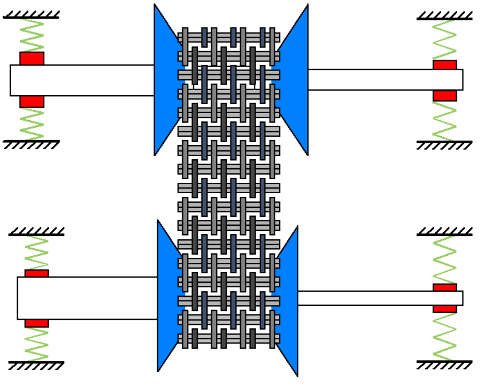


Fig. 8: *Schematic Diagram of Variator Model*

Figure 8 shows an example schematic diagram of the variator model. Using multi-body simulation, all rocker pin elements and links are considered individually. The model includes the influence of the link plate pattern and the pitching, i.e. the sequence of short and long links. The bending of rocker pin elements in the chain is also considered. This bending causes an unequal distribution of link plate forces across the chain width.

The contour of the pulley disks is also a variable. For the contact between the rocker pin element and disk a non-linear contact is applied.

The shaft bearings are applied non-linear and the shaft bending is taken into consideration.

Three main physical loss sources for the calculation of the efficiency are considered:

- Slip between faces of rocker pin elements and pulley disks in circumferential direction and in radial direction
- Rolling losses in the articulations of the rocker pin elements
- Material damping in all components

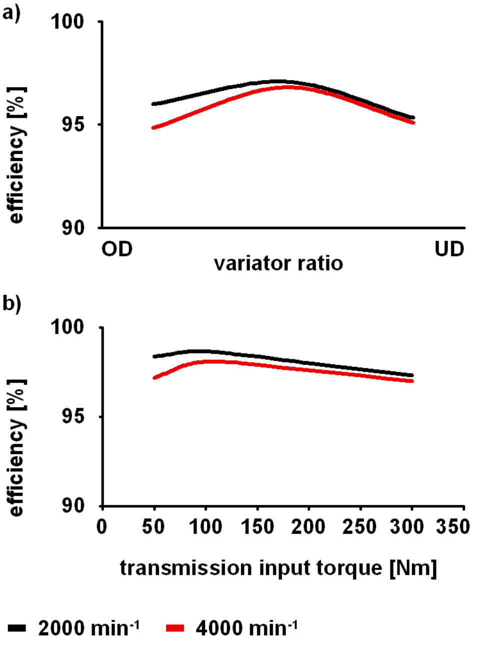
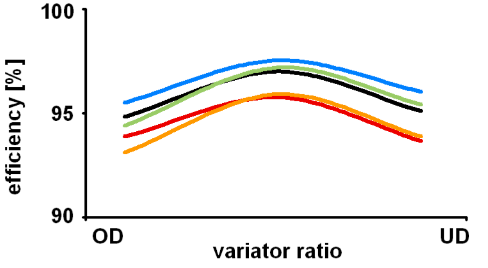


Fig. 9: *Dependency of Efficiency on Ratio (a) and on Engine Torque (b), Calculated with CHAIN Program*



variants		pulley set-geometry	chain	note
▬ (black)	basic A	curved	37 mm	production multitronic®
▬ (green)	B	11°	37 mm	-
▬ (red)	C	7°	37 mm	-
▬ (blue)	D	curved	37 mm ‘light’	shaft: Ø + 2 mm
▬ (orange)	E	curved	37 mm	30% over-clamping

Fig. 10: *Influence of Pulley Set Geometry on Efficiency*

Figure 9 shows the dependency of the efficiency on the variator ratio or transmission input torque, for two different speeds. The indicated theoretical values equate well to the first corresponding measurements.

Geometrically differing variants have been calculated for further optimisation.

Figure 10 shows the corresponding calculation results.

The advantage of the current series (A) compared to version C with 7° pulley set angle is clearly recognisable.

The difference between version B and the current series (A) is minimal. The series version is more efficient in overdrive than version B. This is very advantageous, as a high proportion of all driving across the whole load cycle takes place close to the overdrive range.

In version D, the efficiency increases by around half a percent through increasing the shaft rigidity.

The negative effect of excessive clamping force on efficiency can be clearly seen. As shown in [2], optimising the application system would be expected to improve efficiency by around 2%. This is mainly due to the reduction of losses in the variator and also due to the reduction in the power consumption of the pump.

The reserves in torque capacity shown in the previous chapter can alternatively be used to increase the gear ratio spread. The maximum spread of CVT transmissions are currently around 6.0. Further consumption reductions are to be expected, up to an increase in spread of between 6.5 and 7.0. Ratio spreads beyond this are no longer sensible, as the efficiency drops back again due to the necessary adjustment dynamics and the consequential system design. A spread increase from 6.0 to 6.5 in a middle class vehicle with around 300 Nm engine torque, results in a consumption reduction of around 1%.

Overall, by improving the application system (2%) and using the increased spread (1%), the total consumption can be lowered by a good 3%. A well designed CVT has therefore the potential to also significantly undercut the new 6-speed stepped automatic transmission vehicles in consumption.

Acoustics

A main focus in the CVT development for production use is the acoustic optimisation of the chain and the transmission structure.

Acoustic Optimisation of the Chain

As shown in [1], a proficient distribution of short and long links greatly reduces the monotone consistency of the chain. The pitching is referred to as 'Random' in the following text.

Figure 11 compares a single pitch chain to a random chain. In the artificial head measurements shown, at a constant engine speed the vehicle is continuously decelerated. The illustration shows the frequency over the decreasing vehicle speed. The intensity is shown in colour.

The monotone consistency is clearly recognisable on the single pitch chain, while it disappears with the random chain. However, this change from the tone consistency to noise has caused the level to increase in other areas.

The consecutive impact of rocker pin elements represents the excitation frequency of the chain. Alongside this excitation frequency, the spectrum of the chain acoustics also includes the chain revolution frequency and its multiples. These so-called modulation noises are caused by broad-band excitation of the chain. Along with the reduction of the monotone consistency, as low a modulation noise as possible is the aim.

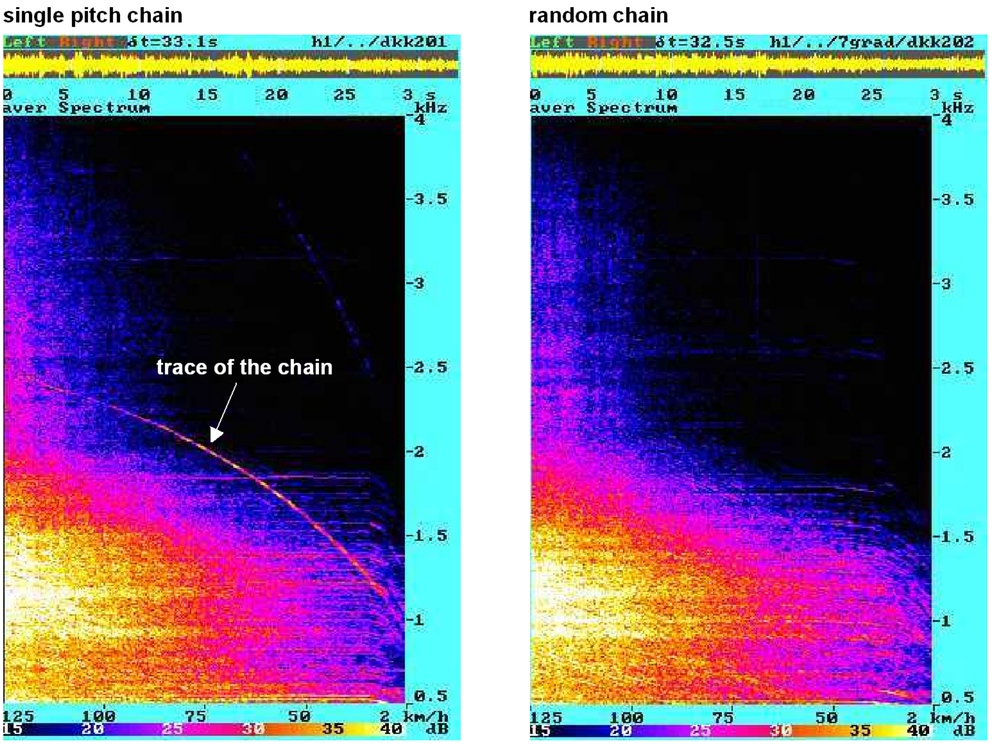


Bild 11: Acoustic Optimisation through Pitching

The chain noise in principle is caused by the impact of the chain links on the pulley sets. Despite the great acoustic success of the random chain, there are also system-dependent optimisation limits for the pitching.

Acoustic Potential of the Transmission Structure

The interruption of the noise transmission paths is an obvious possibility to further improve acoustics. The aim is to interrupt the transmission path as close as possible to the noise source, thus avoiding elaborate secondary measures.

The main transmission path from the noise source to the body is structure-borne noise. The impulse between the chain and the pulleys is carried as structure-borne noise via the pulley set bearings into the transmission housing and continues from there as airborne noise. The obvious aim therefore is to suppress the structure-borne noise at the bearings as much as possible.

Currently elastic bearing couplings are being intensively investigated. To investigate the decoupling effect of the bearings theoretically, it is necessary to analyse the variator's transmission function more thoroughly.

Figure 12 at the top right shows the transmission function of the total variator through a multi-body simulation, on which the excitation is through a single pitch chain. Five main peaks in the transmission function are recognisable, which can be interpreted as structure resonances of the pulley/shaft systems.

By splitting the bearing forces on pulley set 1 and 2, the peaks can be assigned to the corresponding pulley sets.

On pulley set 1 (SS1) two initial bending forms are incited (at 1300 Hz and at 2700 Hz). These are dominant in the underdrive range, which equates to the incitement of a small running radius of the chain.

On pulley set 2 there are three strong resonances (in overdrive range at 1900 Hz and 2200 ... 2800 Hz and also in the underdrive range at 2100 Hz). These three resonance ranges are clearly recognisable in the measurement.

The chain noise is mainly disruptive in the partial-load range. It then makes sense to achieve a dual-stage characteristic curve of the spring for the bearings.

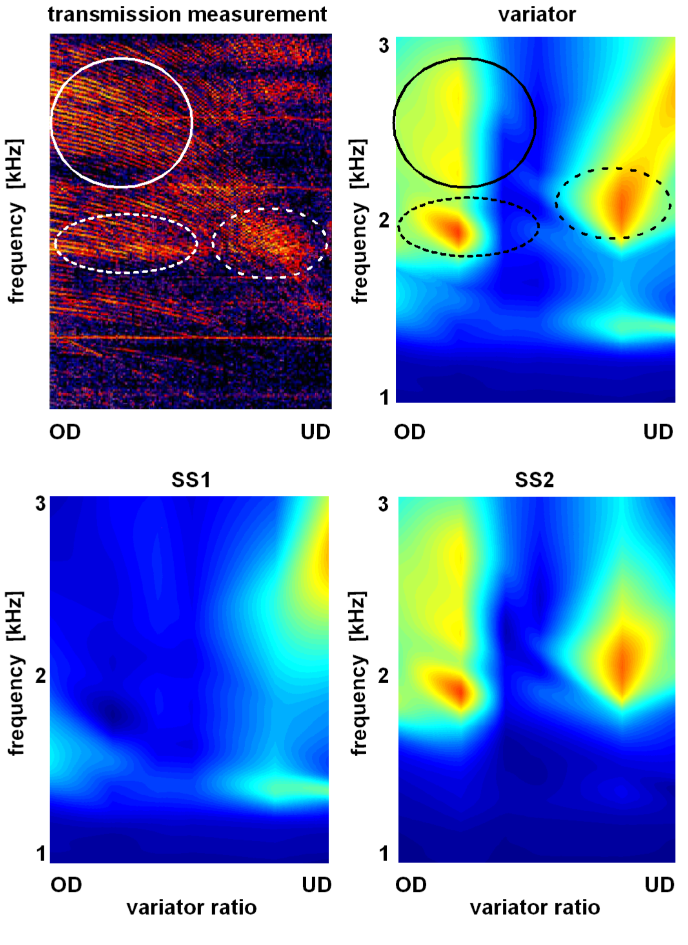


Fig. 12: Transmission Measurement and Three Simulation Results

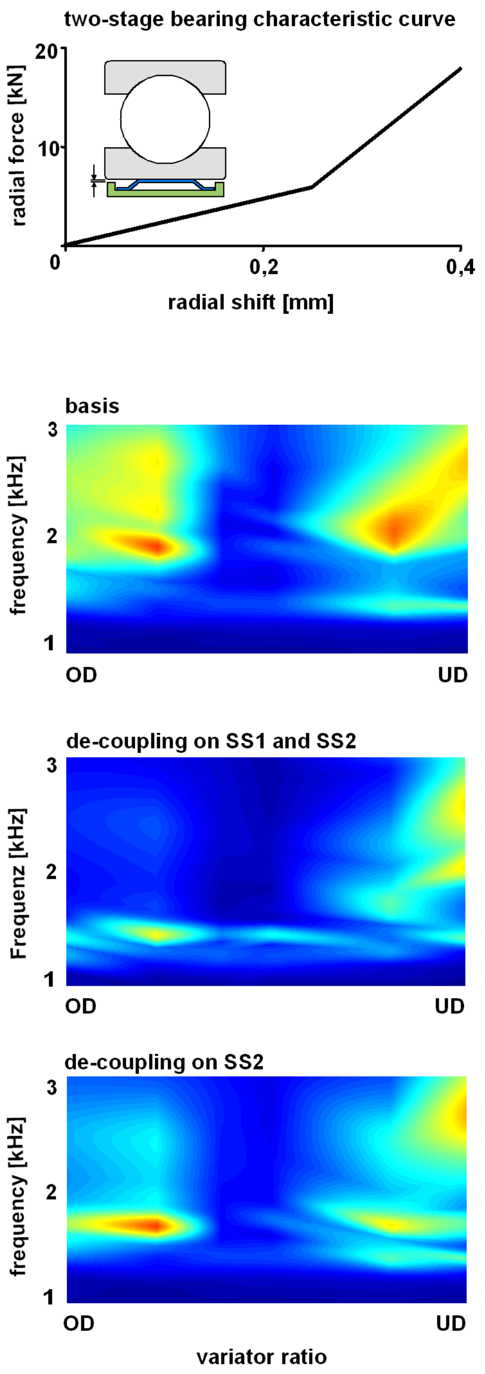


Fig. 13: Characteristic Curves of the Bearings and Simulation Results

Figure 13 shows the assumed bearing characteristic curve and the simulation results.

The decoupling effect of the softer bearing characteristic curves are not only based on the shift of individual resonance frequencies. The acoustically-optimised random chain incites practically all frequencies at all speeds. Therefore, the aim during optimisation must be the increase of the effective damping level in the system.

This is achieved through the reduced system rigidity. However, it is vitally important to avoid the collapse and mutual reinforcement of individual resonances. The best effect is achieved with isolation of pulley set 1 and 2.

Obviously partial constraints exist due to the transmission construction, which will not allow the use of such de-coupled bearings at all bearing locations. The chain variator itself can withstand the de-axialisation relatively well. More critical are shaft areas in which spur gears are arranged.

In co-operation with the company AFT, the effectiveness of such bearing de-coupling has in the meantime been proven on various transmissions.

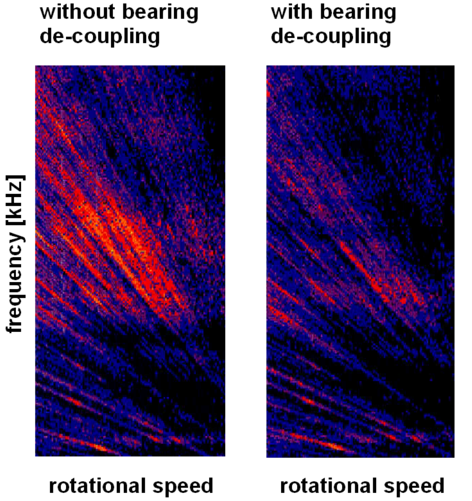


Fig. 14: Measurement of Bearing De-coupling

Figure 14 shows a very good acoustic effect and confirms that this type of noise de-coupling can result in a marked improvement in the transmission noise.

Summary

An essential point of the current development of chain variators is the further increase in torque capacity. Currently it is possible to transmit torques of around 400 Nm with a variator centre distance of approx. 170-180 mm and a transmission spread of 6.0. Multi-range structures, as shown in [3], are a possible means of further increasing torque capacities.

Further progress was made with the optimisation of the efficiency. In this way the CVT transmission can also retain its consumption advantage over manual shift transmissions and the latest 6-speed stepped automatic transmissions.

There are also new approaches to further improve the acoustics. Alongside the previous optimisation of the pitching, acoustic measures on noise transmission paths are also being investigated. The elastic de-coupling between the variator bearings and the housing has proved particularly promising.

The CVT transmission also shows its market viability with regard to cost in comparison with other transmission types.

This then provides all the necessary prerequisites for the CVT transmission to enjoy a successful future on the market.

References

[1] Faust, H.; Linnenbrügger, A.: CVT-Development at LuK, 6th LuK Symposium 1998.

[2] Faust, H.; Homm, M.; Bitzer, F.: Optimising the Efficiency of a CVT Clamping System – Reducing Consumption through Increased Slip, 7th LuK Symposium 2002.

[3] Lauinger, C.; Englisch, A.; Wagner, U.: 500 Nm CVT – LuK Components in Power Split, 7th LuK Symposium 2002.